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यांत्रिक कंपन — नम्य रोटरों के यांत्रिक संतुलन की पद्धतियाँ और मापदंड

Indian Standard

MECHANICAL VIBRATION — METHODS AND CRITERIA FOR THE MECHANICAL BALANCING OF FLEXIBLE ROTORS

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NEW DELHI 110002

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Mechanical Vibration and Shock Sectional Committee, MED 28

NATIONAL FOREWORD

This Indian Standard which is identical with ISO 11342 : 1998 'Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors' issued by the International Organization for Standardization (ISO) was adopted by the Bureau of Indian Standards on the recommendation of the Mechanical Vibration and Shock Sectional Committee and approval of the Mechanical Engineering Division Council.

This standard supersedes IS 14918 : 2001 'Mechanical vibration — Method and criteria for the mechanical balancing of flexible rotors'.

The text of ISO Standard has been approved as suitable for publication as an Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.

b) Comma (,) has been used as a decimal marker in the International Standards, while in Indian Standards, the current practice is to use a point (.) as the decimal marker.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards, which are to be substituted in their respective places, are listed below along with their degree of equivalence for the editions indicated:

International Standard	*Corresponding Indian Standard*	*Degree of Equivalence*
ISO 1925 : 2001 Mechanical vibration — Balancing — Vocabulary	IS/ISO 1925 : 2001 Mechanical vibration — Balancing — Vocabulary	Identical
ISO 1940-1 : 2003 Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances	IS/ISO 1940 (Part 1) : 2003 Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state: Part 1 Specification and verification of balance tolerances	do
ISO 2041 : 1990 Vibration and shock — Vocabulary	IS 11717 : 2000 Vocabulary on vibration and shock (*first revision*)	do
ISO 8821 : 1989 Mechanical vibration — Balancing — Shaft and fitment key convention	IS 14280 : 1995 Mechanical vibration — Balancing — Shaft and fitment key convention	do

The technical committee responsible for the preparation of this standard has reviewed the provisions of following International Standard referred in this adopted standard and has decided it is acceptable for use in conjunction with this standard:

International Standard	*Title*
ISO 1940-2 : 1997	Mechanical vibration — Balancing quality requirements of rigid rotors — Part 2: Balance errors

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Indian Standard

MECHANICAL VIBRATION — METHODS AND CRITERIA FOR THE MECHANICAL BALANCING OF FLEXIBLE ROTORS

1 Scope

This International Standard presents typical flexible rotor configurations in accordance with their characteristics and balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of unbalance, and gives guidance on balance quality criteria.

This International Standard may also be applicable to serve as a basis for more involved investigations, for example when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified methods of manufacture and limits of unbalance, satisfactory running conditions can be expected.

This International Standard is not intended to serve as an acceptance specification for any rotor, but rather to give indications of how to avoid gross deficiencies and/or unnecessarily restrictive requirements.

The subject of structural resonances and modifications thereof is outside the scope of this International Standard.

The methods and criteria given are the result of experience with general industrial machinery. They may not be directly applicable to specialized equipment or to special circumstances. Therefore, there may be cases where deviations from this International Standard may be necessary[1].

2 Normative references

The following standards contain provisions, which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the standards listed below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 1925:1990, *Mechanical vibration — Balancing — Vocabulary*

ISO 1940-1:1986, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 1: Determination of permissible residual unbalance*

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[1] Information on such exceptions will be welcomed and should be communicated to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108/SC1.

ISO 1940-2:1997, *Mechanical vibration — Balancing quality requirements of rigid rotors — Part 2: Balance errors*

ISO 2041:1990, *Vibration and shock — Vocabulary*

ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention*

3 Definitions

For the purposes of this International Standard, the definitions relating to mechanical balancing given in ISO 1925 and the definitions relating to vibration given in ISO 2041 apply.

NOTE — Definitions from ISO 1925 relating to flexible rotors are given for information in annex H.

4 Fundamentals of flexible rotor dynamics and balancing

4.1 General

Flexible rotors normally require multiplane blancing at high speed. Nevertheless, under certain conditions a flexible rotor can also be balanced at low speed. For high-speed balancing two different methods have been formulated for achieving a satisfactory state of balance, namely modal balancing and the influence coefficient approach. The basic theory behind both of these methods and their relative merits are described widely in the literature and therefore no further detailed description will be given here. In most practical balancing applications, the method adopted will normally be a combination of both approaches, often incorporated into a computer package.

4.2 Unbalance distribution

The rotor design and method of construction can significantly influence the magnitude and distribution of unbalance along the rotor axis. Rotors may be machined from a single forging or they may be constructed by fitting several components together. For example, jet engine rotors are constructed by joining many shell, disc and blade components. Generator rotors, however, are usually manufactured from a single forging, but will have additional components fitted. The distribution of unbalance may also be significantly influenced by the presence of large unbalances arising from shrink-fitted discs, couplings, etc.

Since the unbalance distribution along a rotor axis is likely to be random, the distribution along two rotors of identical design will be different. The distribution of unbalance is of greater significance in a flexible rotor than in a rigid rotor because it determines the degree to which any flexural mode is excited. The effect of unbalance at any point along a rotor depends on the mode shapes of the rotor.

The correction of unbalance in transverse planes along a rotor other than those in which the unbalance occurs may induce vibrations at speeds other than that at which the rotor was originally corrected. These vibrations may exceed specified tolerances, particularly at, or near, the flexural critical speeds. Even at the same speed such correction can induce vibrations if the flexural mode shapes on site differ from those dominating during the balancing process.

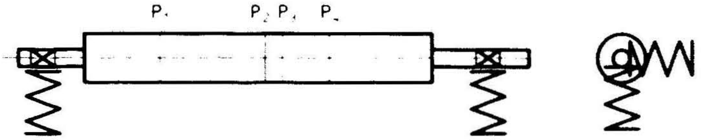
In addition, some rotors which become heated during operation are susceptible to thermal bows which can lead to changes in the unbalance. If the rotor unbalance changes significantly from run to run it may be impossible to balance the rotor within tolerance.

4.3 Flexible rotor mode shapes

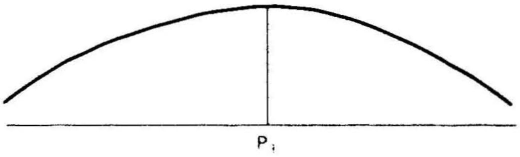
If the effect of damping is neglected, the modes of a rotor are the flexural principal modes and, in the special case of a rotor supported in bearings which have the same stiffness in all radial directions, are rotating plane curves. Typical curves for the three lowest principal modes for a simple rotor supported in flexible bearings near to its ends are illustrated in figure 1.

For a damped rotor/bearing system the flexural modes may be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. Possible damped first and second modes are illustrated in figure 2. In many cases the damped modes can be treated approximately as principal modes and hence regarded as rotating plane curves.

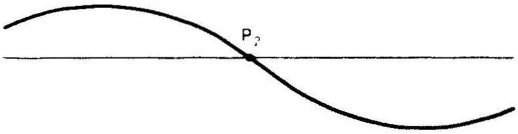
It must be stressed that the form of the mode shapes and the response of the rotor to unbalances are strongly influenced by the dynamic properties and axial locations of the bearings and their supports.



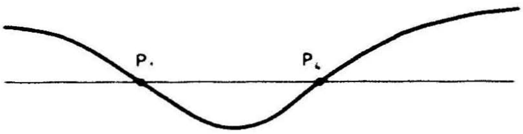
a) Typical rotor



b) First flexural mode



c) Second flexural mode



d) Third flexural mode

NOTE — P_1, P_2, and P_4 are nodes. P_3 is an antinode.

Figure 1 — Simplified mode shapes for flexible rotors on flexible supports

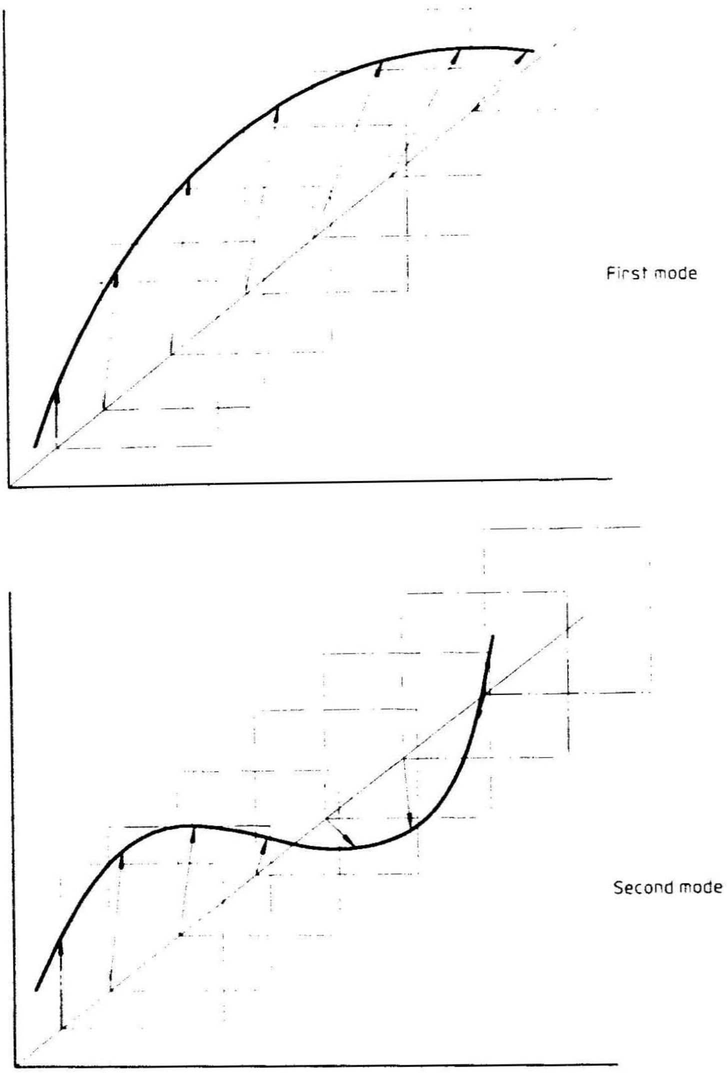


Figure 2 — Examples of possible damped mode shapes

4.4 Response of a flexible rotor to unbalance

The unbalance distribution can be expressed in terms of modal unbalances. The deflection in each mode is caused by the corresponding modal unbalance. When a rotor rotates at a speed near a critical speed, it is usually the mode associated with this critical speed which dominates the deflection of the rotor. The degree to which large amplitudes of rotor deflection occur in these circumstances is influenced mainly by:

a) the magnitude of the modal unbalances;

b) the proximity of the associated critical speeds to the running speeds; and

c) the amount of damping in the rotor/support system.

If a particular modal unbalance is reduced by the addition of a number of discrete correction masses, then the corresponding modal component of deflection is similarly reduced. The reduction of the modal unbalances in this way forms the basis of the balancing procedures described in this International Standard.

The modal unbalances for a given unbalance distribution are a function of the flexible rotor modes. Moreover, for the simplified rotor shown in figure 1, the effect produced in a particular mode by a given correction depends on the ordinate of the mode shape curve at the axial location of the correction: maximum effect near the antinodes, minimum effect near the nodes. Consider an example in which the curves of figure 1 b) to 1 d) are mode shapes for the rotor in figure 1 a). A correction mass in plane P_3 has the maximum effect on the first mode, whilst its effect on the second mode is small.

A correction mass in plane P_2 will produce no response at all on the second mode but will influence both the other modes.

Correction masses in planes P_1 and P_4 will not affect the third mode, but will influence both the other modes.

4.5 Aims of flexible rotor balancing

The aims of balancing are determined by the operational requirements of the machine. Before balancing any particular rotor, it is desirable to decide what balance criteria can be regarded as satisfactory. In this way the balancing process can be made efficient and economical, but still satisfy the needs of the user.

Balancing is intended to achieve acceptable magnitudes of machinery vibration, shaft deflection and forces applied to the bearings caused by unbalance.

The ideal aim in balancing flexible rotors would be to correct the local unbalance occurring at each elemental length by means of unbalance corrections at the element itself. This would result in a rotor in which the centre of mass of each elemental length lies on the shaft axis.

A rotor balanced in this ideal way would have no static and couple unbalance and no modal components of unbalance. Such a perfectly balanced rotor would then run satisfactorily at all speeds in so far as unbalance is concerned.

In practice the necessary reduction in unbalance is usually achieved by adding or removing masses in a limited number of correction planes. There will invariably be some distributed residual unbalance after balancing.

Vibrations or oscillatory forces caused by the residual unbalance must be reduced to acceptable magnitudes over the service speed range. Only in special cases is it sufficient to balance flexible rotors for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, may still experience excessive vibration if it has to run through a critical speed to reach its service speed. However, for passing through critical speeds, the allowable vibration may be greater than that permissible at service speed.

Whatever balancing technique is used, the final goal is to apply unbalance correction distributions to minimize the unbalance effects at all speeds up to the maximum service speed, including start up and shut down and possible overspeed. In meeting this objective, it may be necessary to allow for the influence of modes with critical speeds above the service speed range.

4.6 Provision for correction planes

The exact number of axial locations along the rotor that are needed depends to some extent on the particular balancing procedure which is adopted. For example, centrifugal compressor rotors are sometimes assembly-balanced in the end planes only, after each disc and the shaft have been separately balanced in a low-speed balancing machine. Generally, however, if the speed of the rotor approaches or exceeds its n^{th} flexural critical speed, then at least n and usually $(n + 2)$ correction planes are needed along the rotor.

An adequate number of correction planes at suitable axial positions should be included at the design stage. In practice the number of correction planes is often limited by design considerations and in-field balancing by limitations on accessibility.

4.7 Rotors coupled together

When two rotors are coupled together, the complete unit will have a series of critical speeds and mode shapes. In general, these speeds are neither equal to nor simply related to the critical speeds of the individual, uncoupled rotors. Moreover, the deflection shape of each part of the coupled unit need not be simply related to any mode shape of the corresponding uncoupled rotor. Ideally, therefore, the unbalance distribution along two or more coupled rotors should be evaluated in terms of modal unbalances with respect to the coupled system and not to the modes of the uncoupled rotors.

For practical purposes, in most cases each rotor is balanced separately as an uncoupled shaft and this procedure will normally ensure satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the critical speeds of the uncoupled and coupled rotors, and the distribution of unbalance and the type of coupling and on the bearing arrangement of the shaft train.

If further balancing on site is required, reference should be made to annex A.

5 Rotor configurations

Typical rotor configurations are shown in table 1, their characteristics outlined, and the recommended balancing procedures listed. The table gives concise descriptions of the rotor characteristics. Full descriptions of these characteristics/requirements are given in the corresponding procedures in clauses 6 and 7. The procedures are listed in table 2.

Sometimes a combination of balancing procedures may be advisable. If more than one balancing procedure could be used, they are listed in the sequence of increasing time/cost. Rotors of any configuration can always be balanced at multiple speeds (see 7.3) or sometimes, under special conditions, be balanced at service speed (see 7.4) or at a fixed speed (see 7.5).

Table 1 — Flexible rotors

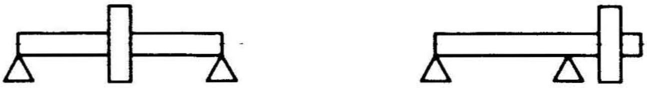
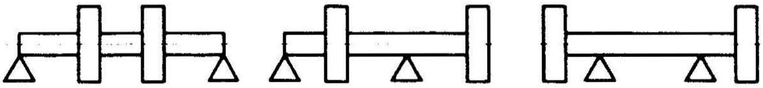
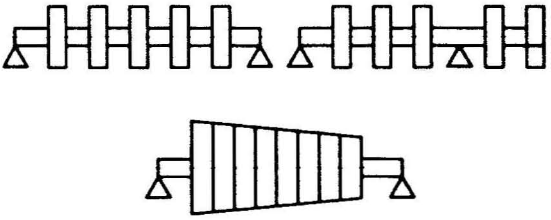
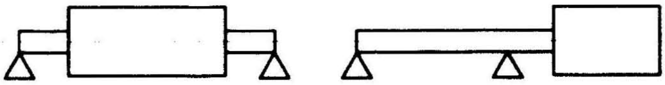
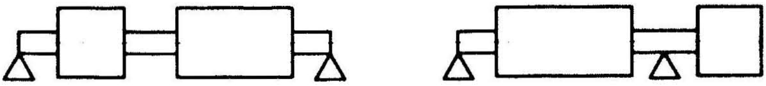
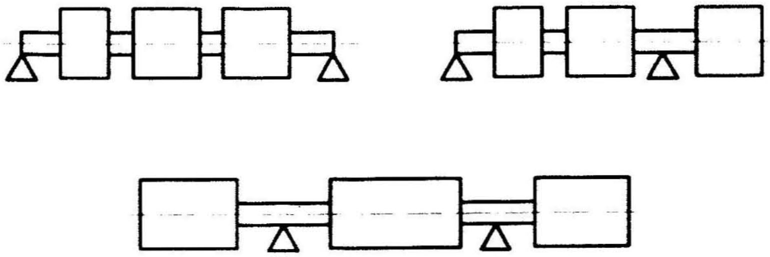
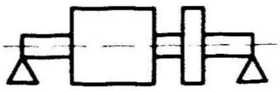
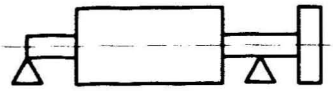
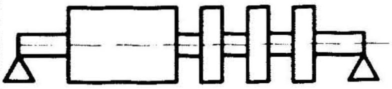
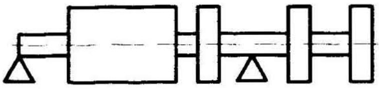
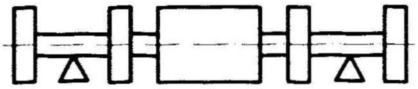
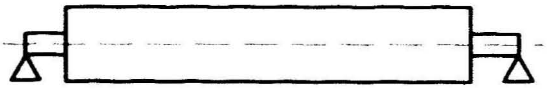
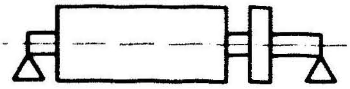
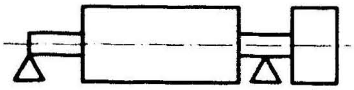
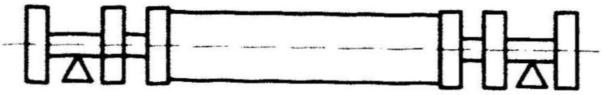
Configuration	Rotor characteristics	Recommended balancing procedure
1.1 Discs	Elastic shaft without unbalance, rigid disc(s)	(see table 2) (see next page for key to A-G)
	**Single disc** - perpendicular to shaft axis - with axial runout	 A; C B, C
	**Two discs** - perpendicular to shaft axis - with axial runout • at least one removable • integral	 B; C B + C, E G
	**More than two discs** - all (but one) removable - integral	 B + C, D, E G
1.2 Rigid sections	Elastic shafts without unbalances, rigid sections	
	**Single rigid section** - removable - integral	 B; C; E B
	**Two rigid sections** - at least one removable - integral	 B + C, E G
	**More (than two) rigid section** - all (but one) removable - integral	 B + C; E G

Table 1 — Flexible rotors (*concluded*)

Configuration	Rotor characteristics	Recommended balancing procedure (see table 2) 1)
1.3 Discs and rigid sections	Elastic shaft without unbalance, rigid discs and sections	
 	**One each** - at least one part removable - integral	B + C; E G
  	**More parts** - all (but one) removable - integral	B + C; E G
1.4 Rolls	Mass, elasticity and unbalance distribution along the rotor	
	- under special conditions - in general	F G
1.5 Rolls and discs/rigid sections	Flexible roll, rigid discs, rigid sections	
 	- discs/rigid sections/removable - under special conditions - in general - integral	 C + F; E + F G G
1.6 Integral rotor	Mass, elasticity and unbalance distribution along the rotor	
	Main parts with unbalances not detachable	G

1) A = Single-plane balancing
B = Two-plane balancing
C = Individual component balancing prior to assembly
E = Balancing in stages during assembly
F = Balancing in optimum planes
G = Multiple speed balancing

Two additional balancing procedures H and I can be used in special circumstances, see 7.4 and 7.5.

Table 2 — Balancing procedures

Procedure	Description	Subclause
	Low-speed balancing	
A	Single-plane balancing	6.5.1
B	Two-plane balancing	6.5.2
C	Individual component balancing prior to assembly	6.5.3
D	Balancing subsequent to controlling initial unbalance	6.5.4
E	Balancing in stages during assembly	6.5.5
F	Balancing in optimum planes	6.5.6
	High-speed balancing	
G	Multiple speed balancing	7.3
H	Service speed balancing	7.4
I	Fixed speed balancing	7.5

6 Procedures for balancing flexible rotors at low speed

6.1 General

Low-speed balancing is generally used for rigid rotors and high-speed balancing is generally used for flexible rotors. However, with the use of appropriate procedures it is possible in some circumstances to balance flexible rotors at low speed so as to ensure satisfactory running when the rotor is installed in its final environment. Otherwise, flexible rotors require use of a high-speed balancing procedure.

Most of the procedures explained in this clause require some information regarding the axial distribution of unbalance.

In some cases where a gross unbalance may occur in a single component, it may be advantageous to balance this component separately before mounting it on the rotor, in addition to carrying out the balancing procedure after it is mounted.

NOTE — Certain rotors contain a number of individual parts which are mounted concentrically (for example blades, coupling bolts, pole pieces, etc.). These parts may be arranged according to their individual mass or mass moment to achieve some or all of the required unbalance correction described in any of the procedures. If these parts need to be assembled after balancing, they should be arranged in balanced sets.

Some rotors are made of individual components (e.g. turbine discs). In these cases it is important to recognize that the assembly process may produce changes in the shaft geometry (e.g. shaft run out) and further changes may occur during high-speed service.

6.2 Selection of correction planes

If the axial positions of the unbalances are known, the correction planes should be provided as closely as possible to these positions. When a rotor is composed of two or more separate components that are distributed axially, there may be more than two transverse planes of unbalance.

6.3 Service speed of the rotor

If the service speed range includes or is close to a flexural critical speed, then low-speed balancing methods should only be used with caution.

6.4 Initial unbalance

The process of balancing a flexible rotor in a low-speed balancing machine is an approximate one. The magnitude and distribution of initial unbalance are major factors determining the degree of success that can be expected.

For rotors in which the axial distribution of initial unbalance is known and appropriate correction planes are available, the permissible initial unbalance is limited only by the amount of correction possible in the correction planes.

For rotors in which the actual distribution of the initial unbalance is not known, there are no generally applicable low-speed balancing methods. However, sometimes the magnitude can be controlled by the prebalancing of individual components. In these cases the low-speed initial unbalance can be used as a measure of the distribution of unbalance.

6.5 Low-speed balancing procedures

6.5.1 Procedure A — Single-plane balancing

If the initial unbalance is principally contained in one transverse plane and the correction is made in this plane, then the rotor will be balanced for all speeds.

6.5.2 Procedure B — Two-plane balancing

If the initial unbalance is principally concentrated in two transverse planes and the corrections are made in these planes, then the rotor will be balanced for all speeds.

If the unbalance in the rotor is distributed within a substantially rigid section of the rotor and the unbalance correction is also made within this section, then the rotor will be balanced for all speeds.

6.5.3 Procedure C — Individual component balancing prior to assembly

Each component, including the shaft, should be low-speed balanced before assembly in accordance with ISO 1940-1. In addition, the concentricities of the shaft diameters or other locating features that position the individual components on the shaft should be held to close tolerances relative to the shaft axis. (See ISO 1940-2).

NOTES

1 The concentricities of the balancing mandrel diameters or other location features that position each individual component on the mandrel should likewise be held within close tolerance relative to the axis of the mandrel. Errors in unbalance and concentricity of the mandrel may be compensated by index balancing (see ISO 1940-2).

2 When balancing the components and the shaft individually, due allowances should be made for any unsymmetrical feature such as keys (see ISO 8821) that form part of the complete rotor but are not used in the individual balancing of the separate components.

3 It is advisable to check by calculation the unbalance produced by balancing errors such as eccentricities and assembly tolerances to evaluate their effects. When calculating the effect of these errors on the mandrel and on the shaft, it is important to note that the effect of the errors can be cumulative on the final assembly. Procedures for dealing with such errors can be found in ISO 1940-2.

6.5.4 Procedure D — Balancing subsequent to controlling initial unbalance

When a rotor is composed of separate components that are balanced individually before assembly (Procedure C), the state of unbalance may still be unsatisfactory. Subsequent balancing of the assembly at low speed is permissible only if the initial unbalance of the assembly does not exceed specified values.

If reliable data on shaft and bearing flexibility, etc. are available, analysis of response to unbalance using mathematical models will be useful.

Experience has shown that symmetrical rotors that conform to the requirements above but have an additional central correction plane may be balanced at low speed with higher initial unbalances of the assembly. Experience has shown that between 30 % and 60 % of the initial resultant unbalance should be corrected in the central plane.

For unsymmetrical rotors that do not conform to the configuration defined above, for example as regards symmetry or overhangs, it may be possible to use a similar procedure using different percentages in the correction planes based on experience.

However, in extreme cases, the initial shaft unbalance may be so large that some other method of balancing the rotor will have to be adopted, for example, Procedure E.

6.5.5 Procedure E — Balancing in stages during assembly

The shaft should first be balanced. The rotor should then be balanced as each component is mounted, correction being made only on the latest component added. This method avoids the necessity for close control of concentricities of the locating diameters or other features that position the individual components on the shaft.

If this method is adopted, it is important to ensure that the balance of the parts of the rotor already treated is not changed by the addition of successive components.

In some cases, it may be possible to add two single-plane components at a time and perform two-plane balancing on the assembly by using one correction plane in each of the two components. In cases where several components form a rigid section, for example a sub-assembly or core section which is normally balanced in two planes only, one such section may be added at a time and corrected by two-plane balancing.

6.5.6 Procedure F — Balancing in optimum planes

If, because of the design or method of construction, a series of rotors has unbalances that are distributed uniformly along their entire length (for example, tubes), it may be possible by selecting suitable axial positions of two correction planes to achieve satisfactory running over the entire speed range by low-speed balancing. It is likely that the optimum position of the two correction planes producing the best overall running conditions can only be determined by experimentation on a number of rotors of similar type.

For a simple rotor system that satisfies conditions a) to e) below, the optimum position for the two correction planes is 22 % of the bearing span inboard of each bearing:

a) single-span rotor with end bearings;

b) uniform mass distribution with no significant overhangs;

c) uniform bending flexibility of the shaft along its length;

d) continuous service speeds not significantly approaching second critical speed;

e) uniform or linear distribution of unbalance.

If this correction method does not produce satisfactory results, it may still be possible to balance the rotor at low speed by utilizing correction planes in the middle and at the rotor ends, as shown in annex B. To do this it is necessary to assess what proportion of the total initial unbalance is to be corrected at the centre plane.

7 Procedures for balancing flexible rotors at high speed

7.1 General

Generally, high-speed balancing is required for flexible rotors. However, with the use of appropriate procedures it is possible, in some circumstances, to balance flexible rotors at low speed (see clause 6).

7.2 Installation for balancing

For balancing purposes, the rotor should be mounted on suitable bearings. In some cases it is desirable that the bearing supports in the balancing facility be chosen to provide similar conditions to those at site so that the modes obtained during site operation will be adequately represented during the balancing process and hence reduce the necessity for subsequent field balancing.

If a rotor has an overhung mass that would normally be supported when installed on site, a steady bearing may be used to limit its deflection during the test.

If a rotor has an overhung mass that is not supported in any way when installed on site, it should also be left unsupported during the test. However, it may be necessary in the early stage of balancing to provide support with a steady bearing to enable a rotor to get safely to service speed or overspeed to allow the rotor components to move into their final position.

Transducers should be positioned to measure shaft, bearing or support vibration or bearing force as appropriate. The system shall be capable of measuring the once-per-revolution component of the signal. The measurement can be expressed either as an amplitude and a phase angle or in terms of orthogonal components relative to some fixed angular reference on a rotor.

In some cases two vibration transducers may be installed 90° apart at the same transverse plane to permit resolution of the transverse vibrations, when such resolution is required.

In all cases, there shall be no resonances of the transducer and/or mountings, which significantly influence vibration measurement within the speed range of the test.

The output from all transducers should be read on equipment that can differentiate between the synchronous component caused by unbalance, the slow-speed runout when significant, and other components of the vibration.

The drive for a rotor should be such as to impose negligible restraint on the vibration of the rotor and introduce negligible unbalance into the system. Alternatively, if known unbalance is introduced by the drive system, then it should be compensated for in the vibration evaluation.

NOTE — To establish that the drive coupling introduces negligible balance error, the coupling should be index balanced as described in ISO 1940-2.

7.3 Procedure G — Multiple-speed balancing

This clause sets out the basic principles of high-speed balancing in a very simple form. The rotor is balanced successively on a modal basis at a series of balancing speeds in turn, which are selected so that there is a balancing speed close to each critical speed within the service speed range. There may also be a balancing speed close to the maximum permissible test speed. In essence, each mode with a critical speed within the service speed range is corrected in turn, followed by a final balance of the remaining (higher) modes at the highest balancing speed.

The procedures used in practice may be packaged in the form of computer-aided balancing methods, which permit automated or otherwise simplified techniques, for example, the influence coefficient method. In the simplest versions, on-line computer-aided balancing will guide the operator through the process and will, for example, perform the vector subtraction listed in 7.3.2.5, 7.3.2.9 and 7.3.2.10. In other cases, prior knowledge of the relevant influence coefficients may be available which can be incorporated in the computer-aided package so that tests with trial mass sets are not required. In appropriate circumstances, vibration data for the unbalanced response can be safely acquired at several balancing speeds during one run of the rotor, rather than at a single balancing speed, so that the necessary corrections for several modes can be computed in one operation.

All vibration (or force) measurements in this clause relate to once-per-revolution components.

7.3.1 Initial low-speed balancing

Experience has shown that it may be advantageous to carry out initial balancing at low speed, prior to balancing at higher speeds. This may be particularly advantageous for rotors significantly affected by only the first flexural critical speed.

If desired, therefore, balance the rotor at low speed, when it is not affected by modal unbalances. Alternatively, this stage can be omitted by proceeding directly to 7.3.2.

NOTE — Low-speed balancing may avoid the need for carrying out the final balancing of the remaining (higher) modes as described in 7.3.2.11.

7.3.2 General procedure

Throughout this procedure, correction planes should be chosen according to the relevant mode shapes. See also clause 4.

7.3.2.1 The rotor should be run at some convenient low speed or speeds to remove any temporary bend. If shaft measuring transducers are used, the remaining repeatable low-speed run-out values should be measured and, where necessary, subtracted vectorially from any subsequent shaft measurements at the balancing speeds.

7.3.2.2 Run the rotor to some safe speed approaching the first flexural critical speed. This will be termed the "first flexural balancing speed".

Record the readings of vibration (or force) under steady-state conditions. Before proceeding, it is essential to confirm that the readings are repeatable. Several runs may be necessary for this purpose.

7.3.2.3 Add a set of trial masses to the rotor, which should be selected and positioned along the rotor to produce a significant vector change in vibration (or force) at the first flexural balancing speed.

If low-speed balancing has been omitted, the trial mass set usually comprises only one mass, which for rotors which are essentially symmetrical about mid-span will be placed near the middle of the rotor span.

If low-speed balancing has been performed, then the trial mass set will usually consist of masses at three distinct correction planes. In this case, the masses are proportioned so that the low-speed (rigid rotor) balancing is not upset.

7.3.2.4 Run the rotor to the same speed and under the same conditions as in 7.3.2.2, and record the new readings of vibration (or force).

7.3.2.5 From the vectorial changes of the readings between 7.3.2.2 and 7.3.2.4, compute the effect of the trial mass set at the first flexural balancing speed. Hence compute the magnitude and angular position of the correction to be applied to cancel the effects of unbalance at the first flexural balancing speed. Add this correction.

NOTES

1 A graphical illustration of the vectorial subtraction underlying this calculation is shown in annex G.

2 In this description it is assumed that the effects on the measurements of unbalances in other modes can be neglected or are eliminated by appropriate procedures.

The rotor should now run through the first flexural critical speed with acceptable vibration (or force). If this is not the case, refine the correction or repeat the procedure in 7.3.2.2 to 7.3.2.5 using a new balancing speed, possibly closer to the first flexural critical speed.

7.3.2.6 Run the rotor to some safe speed approaching the second flexural critical speed. This will be the "second flexural balancing speed". Record readings of vibration (or force) under steady-state conditions at this speed.

7.3.2.7 Add a set of trial masses to the rotor, which should be selected and positioned along the rotor to produce a significant vector change in vibration (or force) at the second flexural balancing speed, without significantly affecting the first mode and, if relevant, the low-speed balance.

7.3.2.8 Run the rotor to the same speed as in 7.3.2.6 and record the new readings of vibration (or force).

7.3.2.9 From the vectorial changes in the readings between 7.3.2.6 and 7.3.2.8, compute the effect of the trial mass set at the second flexural balancing speed for this set of trial masses. Use these values to compute a set of correction masses which cancel the effects of unbalance at the second flexural balancing speed. Attach this set of correction masses.

The rotor should now run through the first and second flexural critical speeds with acceptable vibration (or force). If this is not the case, refine the correction or repeat the procedure in 7.3.2.6 to 7.3.2.9, using a different balancing speed possibly closer to the second flexural critical speed. (See also notes in 7.3.2.5.)

7.3.2.10 Continue the above operations for balancing speeds close to each flexural critical speed in turn within the permissible speed range. Each new set of trial masses should be chosen so that they have a significant effect on the appropriate mode, but do not significantly affect the balance which has already been achieved at lower speeds. The trial mass distribution can be obtained from experience or a computer simulation. For each case, a set of correction masses should be computed and attached to the rotor. Each set of correction masses will compensate for the unbalance at the current balancing speed.

7.3.2.11 If, after correction at all flexural balancing speeds, significant vibrations (or forces) still occur within the service speed range, the procedure in 7.3.2.9 should be repeated at a balancing speed close to the maximum permissible test speed. In this case, it may not be possible to magnify the effect of the remaining (higher) modal unbalance components by running close to their associated flexural critical speeds.

NOTES

1 For some rotor types, for example turbine rotors with shrunk on stages or generator rotors, it is advisable to make only preliminary corrections near the flexural critical speeds to get the rotor to its service speed or overspeed, where components may move into their final position. For some rotors, it may be possible to run safely through some or all of the critical speeds before completing the balancing. In that case, the number of runs required to determine the influence coefficients can be reduced.

2 It should be noted that the method described above assumes that there is a linear relationship between the unbalance vector and the vibration (or force) response vector. In certain cases this may not be so, particularly, for example, where there is a high initial unbalance and the rotor is supported by fluid-film bearings. In these cases it may be necessary to redetermine the effects of the trial mass sets as the vibration (or force) response vector is reduced in magnitude.

3 As explained at the outset of 7.3, the high-speed balancing procedure is presented in a very simple form. In particular, the flexural critical speeds are assumed to be sufficiently widely spaced so that the vibration measured at a flexural balancing speed is predominantly in the mode associated with the corresponding critical speed. If two flexural critical speeds are close together, then more refined procedures (which are beyond the scope of this simple outline) are necessary to uncouple the individual modal components of vibration.

4 For machines that have axial asymmetry (in the support/bearing system), each mode (see figure 1) will split into two modes, often of similar shape, with resonances appearing at different speeds. Reducing the unbalance in one of these modes often reduces the unbalance in the other one too, avoiding the need to balance each mode separately.

7.4 Procedure H — Service-speed balancing

Some rotors that are flexible and pass through one or more critical speeds on their way up to service speed may, under special circumstances, be balanced for one speed only (usually service speed). However, rotors having critical speeds close to service speed or those coupled to other flexible rotors are excluded. In general, these rotors should fulfil one or more of the following conditions:

a) the acceleration and deceleration up to and from service speed is so rapid that the amplitude of vibration at the critical speeds will not build up beyond acceptable limits;

b) the damping of the system is sufficiently high to keep vibrations at the critical speeds within acceptable limits;

c) the rotor is supported in such a manner that objectionable vibrations are avoided;

d) a high level of vibration at the critical speeds is acceptable;

e) a rotor runs at service speed for such long periods that otherwise unacceptable starting/stopping conditions can be tolerated.

A rotor that fulfils any of the above conditions may be balanced in a high-speed balancing machine or equivalent facility at the speed at which it is determined that the rotor should be in balance.

If the rotor falls into category c) above, it is especially important that the stiffness of the balancing machine support system be sufficiently close to site conditions to ensure that, at service speed in the balancing facility, the predominant modes are the same as those that will be experienced at site.

Some consideration should be given to the axial correction mass distribution. It may be possible to choose optimum axial positions for the correction planes so that two planes may be sufficient. This may produce a minimum residual unbalance in the lower modes and thus minimize the vibrations when running through critical speeds.

7.5 Procedure I — Fixed speed balancing

7.5.1 General

These rotors may have a basic shaft and body construction that either allows for low-speed balancing or requires high-speed balancing procedures. In addition, they have one or more components that are either flexible or are flexibly mounted so that the unbalance of the whole system may change with speed.

Rotors in this case may fall into two categories:

a) rotors whose unbalance changes continuously with speed, for example, rubber-bladed fans;

b) rotors whose unbalance changes up to a certain speed and remains constant above that speed, for example rotors of single-phase induction motors with a centrifugal starting switch.

7.5.2 Procedure

It is sometimes possible to balance these rotors with counterbalances of similar characteristics. If not, the following procedures should be used.

Rotors that fall into category a) should be balanced in a balancing machine at the speed at which it is specified that the rotor should be in balance.

Rotors that fall into category b) should be balanced at a speed above that at which the unbalance ceases to change.

NOTE — It may be possible to minimize or counterbalance the effects of the flexible components by careful design and by attention to their locations, but it should be appreciated that rotors of this kind are likely to be in balance at one speed only or within a limited range of speed.

8 Evaluation criteria

8.1 Choice of criteria

It is a usual practice when evaluating the balance quality of a flexible rotor in the factory to consider the once-per-revolution vibration of the bearing pedestals or shaft in a balancing facility or test-bed that reasonably approximates to the site conditions for which the rotor is intended (see however 8.2.4). This is the method described in 8.2.

Another practice is to evaluate the balance quality by considering the unbalance remaining. This is the method described in 8.3. For flexible rotors balanced using low-speed balancing procedures (Procedures A to F), this form of assessment may be made at low speed, without any necessity to use a high-speed balancing facility.

When employing either practice, it is sometimes possible, based on experience, to adjust acceptance levels to permit the use of facilities or installations that do not closely imitate site conditions and/or allow for the final effect of coupling to another rotor on site.

Evaluation criteria are, therefore, established either in terms of vibration limits or permissible residual unbalances.

It is not possible to derive the permissible unbalances for flexible rotors directly from existing documents concerned with the assessment of vibrations in rotating machinery. Usually there is no simple relationship between rotor unbalance and machine vibration under service conditions. The amplitude of vibrations is influenced by many factors, such as the vibrating mass of the machine casing and its foundations, the stiffness of the bearing and of the foundation, the proximity of the service speed to the various resonance frequencies, and the damping.

NOTE — See also ISO 10814.

8.2 Vibration limits in the balancing facility

If the final state of unbalance is to be evaluated in terms of vibration criteria in the balancing facility, then these must be chosen to ensure that the relevant vibration limits are satisfied on site.

There is a complex relationship between vibrations measured in the balancing facility and those obtained in the fully assembled machine at site, which is dependent on a number of factors. It should be noted that acceptance of machines on site is usually based on vibration criteria given in, for example, ISO 7919 or ISO 10816. In most cases this relationship has been derived for specific machine types by experience of balancing typical rotors in the same facility. Where such experience exists it should be used as the basis for defining the permissible vibration in the balancing facility.

There may, however, be cases where such experience does not exist (e.g. a new balancing facility or rotors of substantially different design). Subclause 8.2.5 relates to such cases and explains the permissible levels of once-per-revolution vibration which can be derived from the vibration severity specified in the product specification. If no product specification describing the acceptable running conditions on site exists, reference should be made as appropriate to ISO 7919 or ISO 10816.

8.2.1 General

Numerical values derived according to clause 8 are not intended to serve as acceptance specifications but as guidelines. When used in this manner, gross deficiencies or unrealistic requirements may be avoided.

If due regard is paid to the recommended values, satisfactory running conditions can be expected. However, there may be cases when deviations from these recommendations become necessary.

These recommendations may also serve as the basis for more detailed investigations, for example when, in special cases, a more exact determination of the required balance quality is necessary.

8.2.2 Special cases and exceptions

There are exceptional cases where machinery is designed for special purposes and, of necessity, embodies features which inherently affect the vibration characteristics. Aircraft jet engines and derivatives of such engines for industrial purposes are one example. As engines of this type are designed to minimize weight, their main structures and bearing supports are considerably more flexible than in general industrial machinery. Special steps are taken in the design to accommodate undesirable effects resulting from such support flexibility, and extensive development testing is carried out to ensure that the vibration levels are safe and acceptable for the intended use of the engine.

For such cases as this, where the vibration characteristics have been shown to be acceptable by extensive testing before production units are delivered, it is not intended that the recommendations of clause 8 should apply.

8.2.3 Factors influencing machine vibration

The vibration resulting from the unbalance of the rotor is influenced by many factors such as the mounting of the machine and the distortion of the rotor.

Where maximum permissible levels of vibration are stated in product specifications, they usually refer to total vibration *in situ* arising from all sources. The value quoted could therefore include the vibrations arising from a multiplicity of sources with different frequencies, and the manufacturer should consider what levels of vibration can be permitted from unbalance alone in order to keep within the permissible overall level of vibration.

8.2.4 Critical clearances and complex machine systems

Special attention should be paid to the levels of vibration and static displacement occurring at points of minimum clearance, for example at process fluid seals, because of the greater likelihood of damage at these points than at others. It should be appreciated that the conditions on site may modify the mode shapes and thus the vibration levels at the points of measurements. (See 4.3.)

Rotors that are to be assembled in rigidly coupled multi-bearing systems, for example steam turbine sets, need particular consideration in this respect. The magnitude of the unbalance and its distribution are important factors in such applications. (See annex A.)

8.2.5 Permissible vibrations in the balancing facility

Permissible vibration in the balancing facility can be expressed in two ways:

a) vibration on the bearing pedestal calculated from the permissible bearing vibration on site; or

b) shaft vibration calculated from the permissible shaft vibration on site.

In either case the process[2] can be expressed as:

$$Y = X \times K_0 \times K_1 \times K_2$$

where

X is the permissible total bearing or shaft vibration in the transverse horizontal or vertical direction for measurements taken on site in the service speed range as given in the product specification or the appropriate standard (e.g. ISO 7919 or ISO 10816);

Y is the corresponding permissible once-per-revolution bearing pedestal or shaft vibration in the balancing facility;

K_0 is the ratio of the permissible once-per-revolution vibration to the permissible total vibration ($K_0 \leq 1$);

K_1 is a conversion factor used if the rotor support and/or coupling systems differ from site conditions; it is defined as the ratio of the once-per-revolution measurements in the balancing facility (shaft and/or bearing pedestals) to similar measurements taken on the assembled machine on site (if not applicable, $K_1 = 1$);

K_2 is a conversion factor which is used, if in the balancing facility shaft measurements are taken at locations other than those for which X is specified. Its value depends on the modal characteristics of the rotor. If the measurement locations are the same, then $K_2 = 1$.

NOTES

1 The conversion relationship gives units for Y which are the same as those for X. In practice it may subsequently be convenient to express Y in different units. For example, displacement instead of velocity.

2 The value of K_1 will often depend upon the direction of measurement.

3 For cases in which the measurement cannot be made at the same locations, K_2 may be determined analytically using a rotor dynamics model of the system.

The values of K_1 and K_2 may vary widely between one installation and another and will be speed dependent. Some suggested values for K_0 and K_1 are shown in annex C. The value of K_2 needs to be established for each specific application. If a critical speed of a particular configuration of the rotor bearing system coincides with the service speed, higher values of the relevant conversion factors have to be used.

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[2] Users are recommended to compare the above process with their own experience. Comments on the results will be welcome and should be directed to the national standards body in the country of origin for transmission to the Secretariat of ISO/TC108/SC1.

It should be noted that in practice it is not essential that these conversion factors are determined in isolation, provided that a composite factor is available.

In addition, it should be noted that modal amplification of the vibration will occur at critical speeds. Balancing practice is therefore usually directed not only towards satisfactory limitation of vibration within the service speed range, but also towards smooth passage through critical speeds below the maximum service speed. For critical speeds it is especially difficult to establish quantitative criteria because it is almost impossible to arrange the same support conditions in the balancing facility as *in situ* (especially damping).

When the bending deflection during run up is of concern, because of rotor/stator clearances or stresses, the bending of a rotor at critical speeds below service speed should be considered in terms of displacement of that part of the rotor at which the displacement is of consequence.

8.3 Residual unbalance limits

This subclause provides recommendations for rigid rotor unbalance and modal unbalances for a flexible rotor based on criteria given in ISO 1940-1.

8.3.1 General

The following establishes guidelines for the required balance quality of flexible rotors. The values given are based on a limited amount of documented practical experience with the various types of rotor. However, if due regard is paid to the recommended values, satisfactory running conditions can be expected. Nonetheless, the suggested levels and classifications are not yet completely verified, and deviations from these recommendations may be necessary in certain cases[3].

For flexible rotors balanced at low speed, permissible residual unbalances in specified correction planes are used to state the balance quality. For rotors balanced at high speed, permissible residual modal unbalances are applied.

8.3.2 Limits for low-speed balancing

The residual unbalance for any completely assembled rotor should not exceed the residual unbalance recommended for an equivalent rigid rotor in ISO 1940-1.

In addition, for rotors which are balanced in accordance with procedure C, D or E (see table 2) each component, or when applicable, each sub-assembly of components should be balanced to limits based on experience or those recommended in ISO 1940-1, applied to each component.

8.3.3 Limits for multiple-speed balancing

8.3.3.1 First bending mode

For a rotor that is significantly affected by only the first modal residual unbalance, then whatever its unbalance distribution, the residual unbalance should not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid rotor in ISO 1940-1 and based upon the highest service speed of the rotor:

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[3] Reports of any such deviations will be welcome. Comments should be directed to the National Standards body in the country of origin for transmission to the secretariat of ISO/TC 108/SC 1 and will be taken into account when preparing subsequent editions of this International Standard.

a) the equivalent first modal residual unbalance should not exceed 60 %; and

b) if low-speed balancing is carried out initially, the total residual unbalance as a rigid rotor should not exceed 100 %;

c) the residual unbalance at service speed should not exceed 100 % (see 9.2.3 for guidance).

8.3.3.2 First and second bending modes

For a rotor that is significantly affected by only the first and second modal unbalances, then whatever its unbalance distribution, the residual unbalance should not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid rotor in ISO 1940-1 and based upon the highest service speed of the rotor:

a) the equivalent first modal residual unbalance should not exceed 60 %; and

b) the equivalent second modal residual unbalance should not exceed 60 %; and

c) if low-speed balancing is carried out initially, the total residual unbalance as a rigid rotor should not exceed 100 %;

d) the residual unbalance at service speed should not exceed 100 % (see 9.2.3 for guidance).

In cases when one of the modes is less significant than the other, the corresponding limit can be relaxed, but shall not exceed 100 %;

NOTE — The example in annex F illustrates the calculation of these limits.

8.3.3.3 More than two bending modes

For rotors which are significantly affected by more than the first and second modal unbalance, no recommendations are available.

The following notes relate to 8.3.

NOTES

1 A method for the experimental determination of the equivalent modal residual unbalances is described in 9.2.2.

2 If the influence of overhung masses is significant, then the percentages given may not be applicable.

3 If, *in situ*, the service speed or service speed range is close to either the first or second flexural critical speed, these figures may require modification.

4 In the balancing facility, the proposed limits will not necessarily result in vibration magnitudes within normal limits in the speed range from 80 % to 120 % of any critical speed. If such amplified vibrations occur, it does not necessarily mean that more refined balancing is needed because, for example, damping in the balancing facility is often smaller than *in situ*.

5 When all relevant rotor flexural modes cannot be taken into account in the balancing facility (for example, due to an insufficient number of correction planes), a decision should be reached concerning which modes should be emphasized for balancing.

9 Evaluation procedures

Depending on the type and purpose of the rotor being assessed, the final state of unbalance may be evaluated either in terms of vibration at specified measuring planes, or by residual unbalances.

NOTE — In cases of small mass-produced rotors, assessment procedures simpler than those detailed in this International Standard may suffice.

9.1 Evaluation procedures based on vibration limits

9.1.1 Vibrations assessed in a high-speed balancing facility

The installation of the rotor in the test facility should conform to the guidelines given in 7.2.

When the above conditions have been satisfied, the rotor should be run up to speed at a low acceleration rate to ensure that vibration peaks are not suppressed. If measurement over the whole speed range is not possible, all significant peaks of vibration should be measured between 70 % of the observed first flexural critical speed and the maximum service speed. Alternatively, this could be achieved by a comparable run down.

The rotor should be held at maximum service speed long enough to eliminate any transient effects. Synchronous vibration measurements should then be taken.

9.1.2 Vibrations assessed on the test bed

Rotors whose final state of unbalance is evaluated on the test bed should have instrumentation as stated in 7.2, but it should be noted that different procedures may be necessary in some cases when, for example:

a) the rotor is assembled as a complete machine driven by its own power;

b) only full-speed readings can be obtained, such as for an induction motor;

c) vibration transducers cannot be placed at the bearings; in this case the points where vibrations should be measured should be agreed between the manufacturer and the user;

d) the state of unbalance depends on load, in which case the range of load over which the residual unbalance is assessed should be agreed between the manufacturer and the user.

9.1.3 Vibrations assessed at site

9.1.3.1 Machines that have their state of balance assessed after final installation at site are subject to many factors that can produce vibration. Some of this vibration may be at shaft rotational frequency from sources other than mechanical unbalance. Some of the factors that can produce such vibrations, together with some of the precautions that should be taken, are mentioned in annex A.

9.1.3.2 If any of the stationary parts of the machine or the supporting foundation structure are in resonance at the service speed, high levels of vibration are sometimes produced even though the rotor residual unbalance is well within normally accepted tolerances.

In such circumstances, balancing within exceptionally fine limits may be required to reduce the vibration level. Such improvements may be only useful if the machine is not highly susceptible to unbalance. If, in operation, there is a high probability that new unbalances will occur, consideration should be given to the practicality of eliminating the structural resonances or increasing the damping in the system or other measures, so that satisfactory operation can be obtained.

9.1.3.3 In the final installation at site, there may be factors during commissioning that will conflict with obtaining the steady-state conditions needed to assess the state of balance. It may then be necessary to combine the result of balancing runs with tests for other purposes. If the preliminary running of the installed machine shows the result of balancing to be in doubt, special runs should be arranged specifically for confirming the adequacy of the balance.

In many installations (for example where the prime mover is a 'direct to line start' induction motor), it may be impossible to control the speed of rotation during run up and steady conditions can only be achieved at full speed.

Agreement should therefore be reached between the manufacturer and user on the speed range over which the state of balance should be checked.

The balance check is normally made with the machine unloaded. If the machine is loaded, the load at which the state of balance is to be checked should be agreed between the manufacturer and user.

9.1.3.4 Vibration measuring equipment should be installed as specified in 7.2. Where suitable monitoring equipment is provided in the installation, this may be used instead. Alternatively, vibrations may be read on portable apparatus using, for instance, a hand-held vibration transducer.

9.2 Evaluation based on residual unbalance limits

Three different approaches are outlined below.

9.2.1 Evaluation at low speed

The evaluation at low speed is based on the limits for rigid rotors as stated in ISO 1940-1.

Rotors in this category usually have their balance quality assessed in a low-speed balancing machine. In most cases a subsequent high-speed check will be made on the test bed or on site. In specific cases, by agreement between the manufacturer and user, the high speed assessment may be dispensed with and the rotor accepted on the basis of the residual unbalance at low speed. This applies particularly to rotors sold as spares where a final assessment at site may be delayed for a considerable time.

The rotor should be complete and all attachments such as half couplings, gear wheels, etc., should be fitted.

The balancing machine should be one that conforms to ISO 2953. See ISO 1940-1 and ISO 1940-2 for the procedure for assessing residual unbalance and cautionary comments.

Before the residual unbalance of the rotor is assessed, it should be run at some suitable speed to remove any temporary bend.

When the above conditions have been satisfied, the rotor should be run at the balancing speed and readings taken of amount and angle of unbalance remaining in each measuring plane.

For rotors with controlled initial unbalance, the initial unbalance after assembly should also be stated in addition to the measured residual unbalance. For rotors that have been balanced in several stages during assembly or that have been made up of balanced components (Procedure E), the residual unbalance achieved at each stage should be stated.

9.2.2 Evaluation at multiple speeds based on modal unbalances

Multiple speeds will give an insight into the unbalance distribution of the rotor and its expected flexible behaviour.

To assess the state of unbalance, the residual equivalent modal unbalances are calculated for the respective modes. The equivalent modal unbalance is defined as the smallest unbalance in an individual plane which has the same effect as the modal unbalance (see definition, annex H). This means that, for each respective mode, the residual unbalance is calculated for the most sensitive plane. This assumes, that balancing planes are located in suitable positions.

The procedure is as follows.

a) Mount the rotor in a high-speed balancing machine or other high-speed test facility.

b) If low-speed balancing is performed, the residual unbalance in the rigid rotor state may be assessed either by using the influence coefficient method, or by using the balancing machine and its capability to indicate unbalances in two planes.

c) Run the rotor to some safe speed approaching first flexural critical speed and note readings of bearing vibrations or forces.

d) Add a trial unbalance to the rotor. The unbalance should be sufficient to show a significant effect and should be placed axially where it will have the maximum effect on first mode. Take readings of bearing vibration or forces at the same speed as in c).

e) From the readings obtained in c) and d), compute vectorially the equivalent first modal unbalance. For example, this can be done graphically by the construction in annex G, in this case with the single unbalance mass forming the trial mass set. The magnitude of the equivalent first modal unbalance is:

$$\mathrm{TU} \times \frac{\mathrm{AO}}{\mathrm{AB}}$$

where TU is the trial unbalance.

f) Remove the trial unbalance.

g) Run the rotor to some safe speed approaching second flexural critical speed, provided this is lower than the maximum safe service speed. Note readings of bearing vibrations or forces.

h) Add a trial unbalance to the rotor. This should be sufficient to show a significant effect and should be placed axially where it will have maximum effect on second mode. Take readings of bearing vibrations or forces at the same speed as in g).

i) From the readings obtained in g) and h) compute vectorially the equivalent second modal unbalance. The graphical procedure of e) may be used in this case.

j) Remove the trial unbalance.

k) Continue the operations for successive modes until the equivalent modal unbalances in all significant modes have been determined.

An example is shown in annex D.

NOTES

1 When determining the equivalent modal unbalances, it may be desirable to use a set of trial unbalances in order to pass safely through the low critical speeds. Finally each calculated residual unbalance set is summarized into a residual equivalent modal unbalance.

2 The procedure given assumes that the vibration measured at a speed close to a critical speed is predominantly in the corresponding mode and, therefore, usually gives a close approximation to the equivalent residual modal unbalances.

3 Sometimes it may not be possible to run close to the critical speeds of some of the significant modes. In these cases further procedures are necessary to separate the individual modal components.

4 If the rotor remains in the balancing facility after a balancing procedure, in accordance with 7.3, the information obtained during balancing may be used directly, without the need for further test runs.

9.2.3 Evaluation at service speed in two specified test planes

If the service speed is used, special care is needed to choose the test planes properly.

The axial position of the correction planes and the balancing speed should be stated.

If the rotor is assessed in a balancing facility having its own instrumentation this should be used throughout the test.

If the rotor is assessed in an overspeed or similar facility, the instrumentation and general installation of the rotor into the facility should be as stated in 7.2.

Annex A

(informative)

Cautionary notes concerning rotors on site

A.1 Introduction

Unbalance is not the only cause of vibration, not even once-per-revolution vibration. Before undertaking balancing or related operations, due consideration should be given to identifying the factors other than unbalance that are influencing the magnitude of vibration of the machine. Such factors include those mentioned below.

This is particularly true in installations where two or more rotors are coupled together, such as turbine generator units.

A.2 Bearing misalignment

Small parallel or angular misalignment of the rotor bearings can produce effects which are not caused by unbalance. If these effects are present, the misalignment may need to be corrected prior to further assessment of the vibration of the machine (see also the last paragraph of A.3).

A.3 Radial and axial runout of coupling faces

There is no practical way of ensuring that large rotors can be coupled together without a small amount of radial and axial runout of the coupling faces between the mating halves of the coupling. These runouts may produce vibration effects which cannot be satisfactorily corrected by balancing. Therefore, if the machine is not responding to balancing operations, the radial and axial runout of the coupling faces should be checked.

Where appropriate, errors should be corrected to lie within tolerances which have been found to be satisfactory in practice for the size and type of machine under consideration, before attempting further balancing.

A.4 Bearing instability

Various forms of instability (for example fluid whirl/whip) may take place in the types of hydrodynamically lubricated bearings which are normally used in multispan flexible rotor systems.

The symptoms of these phenomena are well known, and it is necessary to ascertain whether such symptoms are present before attempting to improve the quality of running by balancing.

Discussions of such effects and possible remedial measures are outside the scope of this International Standard.

Annex B
(informative)

Optimum planes balancing — Low-speed three-plane balancing

B.1 This annex is concerned with the low-speed balancing of rotors which have one central and two end correction planes, if all of the following conditions are met:

a) single-span rotor with no significant overhang;

b) uniform or linear distribution of unbalance;

c) uniform bending flexibility of rotor along its length;

d) symmetrical position of end correction planes about midspan;

e) continuous service speeds below and not significantly approaching second critical speed.

Such rotors can be satisfactorily balanced on a low-speed balancing machine provided that an assessment can be made of the proportion of the total unbalance of the rotor which should be corrected at the central plane. This annex provides a method whereby the balance correction in three planes may be calculate from the initial unbalance measured in two planes. The vector sum of the forces and the vector sum of the moments created by the three unbalance corrections $\vec{U}_1$, $\vec{U}_2$ and $\vec{U}_3$ about a given point on the rotor should compensate those caused by the initial unbalances, $\vec{U}_L$ and $\vec{U}_R$ about the same point.

B.2 It can be shown that the initial unbalance will be completely corrected up to, and including, its first modal component when the following vector relationships are satisfied:

$$\vec{U}_1 = \vec{U}_L - 1/2\,H\left(\vec{U}_L + \vec{U}_R\right)$$
$$\vec{U}_2 = H\left(\vec{U}_L + \vec{U}_R\right)$$
$$\vec{U}_3 = \vec{U}_R - 1/2\,H\left(\vec{U}_L + \vec{U}_R\right)$$

where H is the central correction divided by the initial static unbalance.

It should be noted that $\vec{U}_L$, $\vec{U}_R$, $\vec{U}_1$, $\vec{U}_2$ and $\vec{U}_3$ are vectors.

Values of H are presented graphically in figure B.1 as a function of z/l, where z is the distance from the left-hand bearing to correction plane 1 and l is the bearing span (shaft length).

It should be noted that H is zero when $z/l = 0.22$, which indicates that in this case the centre plane is no longer needed and the procedure has become a two-plane balancing procedure, usually called "quarter-point balancing". For values of z/l greater than 0.22, the correction in the centre plane is on the opposite side of the shaft.

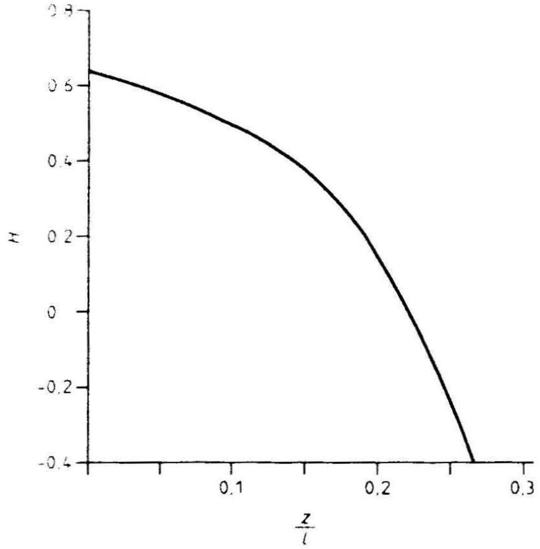

Figure B.1 — Graphical presentation for determination of *H*

Annex C

(informative)

Conversion factors

The machinery classification according to ISO 10816-1 is as follows.

I — Individual parts of engines and machines, integrally connected to the complete machine in its normal operating condition.

II — Medium-size machines without special foundations and rigidly mounted engines or machines (up to 300 kW) on special foundations.

III — Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations that are relatively stiff in the direction of vibration measurement.

IV — Large prime movers and other large machines with rotating masses mounted on foundations that are relatively soft in the direction of vibration measurement.

Table C.1 — Suggested conversion factor ranges (see 8.2.5)

Machinery classification	Typical machines	K_0	K_1		
			Bearing support absolute	**Shaft absolute**	**Shaft relative**
I	Superchargers	1.0	0.6 to 1.6	1.6 to 5.0	1.0 to 3.0
	Small electric motors up to 15 kW	1.0			
II	Paper-making machines	0.7 to 1.0			
	Medium-sized electric machine 15 kW to 75 kW	0.7 to 1.0			
	Electrical machines up to 300 kW on special foundations	0.7 to 1.0			
	Compressors	0.7 to 1.0			
	Small turbines	1.0			
III	Large electric motors	0.7 to 1.0			
	Pumps	0.7 to 1.0			
	2-pole generators	0.8 to 1.0			
	Turbines and multipole generators	0.9 to 1.0			
IV	Gas turbines (but see 8.2.2)	1.0			
	2-pole generators	0.8 to 1.0			
	Turbines and multipole generators	0.9 to 1.0			

K_0 is the ratio of the permissible once-per-revolution vibration to the permissible total vibration ($K_0 \leq 1$).

K_1 is the ratio of the once-per-revolution measurements in the balancing facility (shaft and/or bearing pedestals) to similar measurements taken on the assembled machine on site. (If not applicable, $K_1 = 1$).

NOTE 1 In relation to the entries for K_1, "absolute" refers to measurement of vibration with reference to an inertial reference frame and "relative" refers to measurement relative to an appropriate structure, such as a bearing housing. A full discussion of these terms is given in ISO 7919-1.

NOTE 2 Users are recommended to compare the above values with their own experience. Comments on the results or such comparisons will be welcome and should be directed to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108/SC 1.

Annex D

(informative)

Example — Calculation of equivalent modal residual unbalance

NOTE — A recommended procedure is outlined in 9.2.2.

D.1 Residual unbalance calculation

The principles of residual unbalance calculation are given in the example below. The rotor is a turbine rotor with four balancing planes (see figure D.1). The balancing calculations are based on the vibration measurements at the two bearings (transducer 1 and transducer 2).

The service speed of the rotor is 10 125 r/min.

Rotor mass is 1625 kg.

Permissible total unbalance for an equivalent rigid rotor according to G2.5 (2,37 g·mm/kg) from ISO 1940-1.

Total residual unbalance for an equivalent rigid rotor.

$$2{,}37\ \frac{\text{g}\cdot\text{mm}}{\text{kg}} \times 1\,625\ \text{kg} = 3\,850\ \text{g}\cdot\text{mm}$$

Permissible equivalent first modal unbalance (100 %)..... 3 850 g·mm

NOTE — Since the service speed is above the second critical speed, a factor of 100 % has been taken instead of 60 % of the first modal unbalance; see note in 8.3.3.2.

Permissible equivalent second modal unbalance (60 %)..... 2 311 g·mm

Total permissible residual unbalance for the rigid
rotor (low-speed balancing) ..... 3 850 g·mm
(1 925 g·mm per plane Bp1 and Bp3)

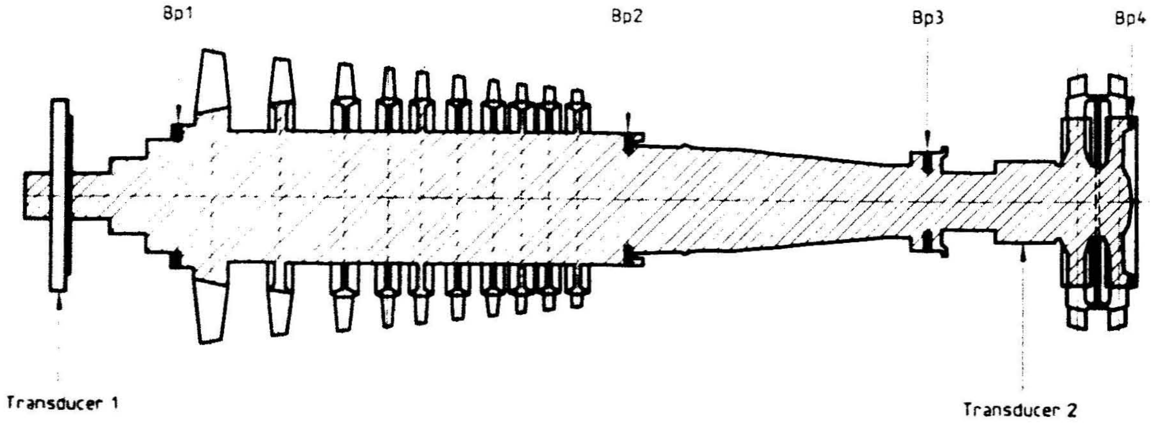

Figure D.1 — Turbine rotor

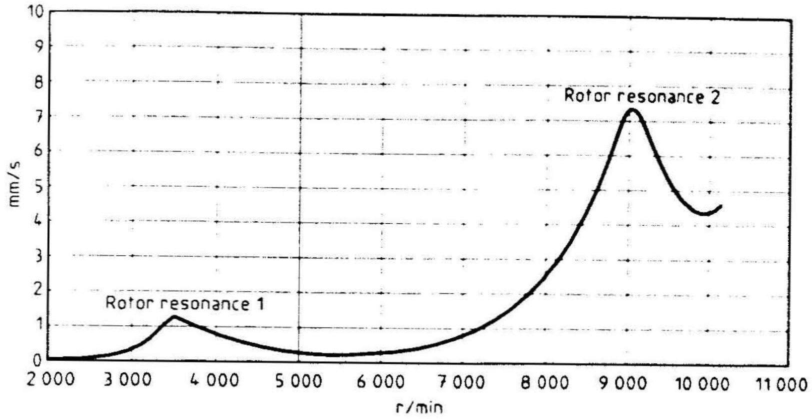

Figure D.2 — Run-up curve — Before balancing

D.2 Influence coefficients

The balancing speeds for this rotor are:

1 000 r/min (low speed)
3 400 r/min (just below rotor resonance 1)
9 000 r/min (just below rotor resonance 2)

The influence coefficients given in table D.1 have been calculated from runs with trial masses.

Table D.1

Measurement point	**Balancing plane**				**Speed**
	Bp1	Bp2	Bp3	Bp4	
Transducer 1	*0,0594/3°	0,0330/1°	*0,00912/333°	0,00490/233°	1000
Transducer 2	*0,00216/35°	0,0227/14°	*0,0334/11°	0,0425/9°	r/min
Transducer 1	0,249/82°	0,343/94°	0,055/222°	*0,360/265°	3400
Transducer 2	0,087/107°	0,157/87°	0,102/34°	*0,224/6°	r/min
Transducer 1	1,99/246°	*2,29/285°	1,56/293°	2,07/176°	9000
Transducer 2	1,92/353°	*1,99/134°	1,16/109°	0,595/281°	r/min

The influence coefficients are given in the unit (mm/s)/(kg·mm) and at an angle relative to a reference system on the rotor.

The influence coefficients used for residual unbalance calculation are marked with an asterisk. Bp1 and Bp3, which are nearest to the bearings, are selected for the low speed. For other speeds, the most sensitive plane for each transducer is selected.

D.3 Final vibration readings

The vibrations given in table D.2 were measured during the run in the final balance condition.

Table D.2

Speed	Transducer 1	Transducer 2	Unit
1 000 r/min	0,01/237°	0,022/147°	mm/s
3 400 r/min	0,55/52°	0,22/125°	mm/s
9 000 r/min	2,35/305°	1,44/139°	mm/s

D.4 Residual unbalance at low speed, 1 000 r/min

The calculation is carried out in accordance with the influence coefficient method for correction planes Bp1, Bp3 (planes nearest to the bearings) and transducers 1 and 2.

Table D.3

	Calculated	Permissible
Bp1	246 g·mm	1925 g·mm
Bp3	671 g·mm	1925 g·mm

At the other balancing speeds, the residual unbalance is obtained by dividing the amplitude of the vibrations by the absolute value of the coefficient. This means that no consideration need be given to the phase angle of the vibrations or the phase information of the receptance.

D.5 Residual unbalance at 3 400 r/min

Table D.4

	Calculated	Permissible
Transducer 1	(0,55/0,36) 1000 = 1530 g·mm	3850 g·mm
Transducer 2	(0,22/0,224) 1000 = 982 g·mm	3850 g·mm

D.6 Residual unbalance at 9 000 r/min

Table D.5

	Calculated	Permissible
Transducer 1	(2,35/2,29) 1000 = 1026 g·mm	2311 g·mm
Transducer 2	(1,44/1,99) 1000 = 723 g·mm	2311 g·mm

Annex E

(informative)

Procedure to determine if a rotor is rigid or flexible

E.1 General

This annex describes procedures that may be used to determine whether a rotor is rigid or flexible. If it is determined that a rotor falls into the rigid category, then it can be balanced using a low-speed balancing procedure. In general, flexible rotors need to be balanced at high speeds using procedures such as those in clause 7. There are, however, rotors which, by definition, are flexible but are borderline, and for which low-speed balancing may be adequate using the special procedures given in clause 6.

The physical appearance of a rotor is insufficient to determine whether a rotor falls into the rigid or flexible categories for balancing purposes. If a rotor operates at high speeds, it may approach or pass through a critical speed involving significant bending of the rotor and therefore requires high speed balancing. A rotor is considered rigid for balancing purposes if its highest service speed is at least 30 % below the first flexural critical speed.

E.2 Determination of whether a rotor is rigid or flexible

One or more of the following may be used to ascertain whether a rotor is rigid or flexible, and thereby determine the balancing method to be adopted.

E.2.1 Consult the rotor manufacturer or the user for a definition of the rotor configuration and characteristic and a recommended balancing procedure. (See clause 5.)

E.2.2 If the first flexural critical speed exceeds the maximum service speed by at least 50 %, then the rotor can often be considered rigid for balancing purposes.

E.2.3 Alternatively the following test sequence may be performed.

Balance the rotor at low speed in two correction planes in accordance with the procedures specified in ISO 1940.

Mount the rotor in a facility that is capable of rotating the rotor to at least service speed and that has stiffness and damping of the bearings and their supports similar to the service installation. Accelerate the rotor gradually to service speed, taking care that vibration at all times stays within safe limits. Record vibration readings as a function of speed during the acceleration and subsequent deceleration.

If no significant change in vibration occurs as a function of speed, then the rotor is either rigid, or is flexible with low levels of modal unbalance. To determine which of these possibilities apply, perform the flexibility test defined in E.3.

If a significant change in vibration occurs during acceleration or deceleration, one or more of the following possibilities exist:

a) the rotor is flexible;

b) the rotor is rigid but flexibly supported;

c) the rotor has components that shift location significantly as a function of speed or temperature.

To help discriminate between these possibilities, accelerate the rotor again to service speed and then check if the readings during deceleration to zero speed repeat those of the prior deceleration run. If they do the rotor has settled. Next perform the flexibility test given in E.3 to determine if the rotor is rigid or flexible.

NOTE — Settlement may have occurred by taking the rotor to its service speed or beyond by permanently 'seating' components due to centrifugal force. For example, generator and motor rotors frequently require a settlement run to enable the copper windings and support systems to move radially outward to their final position.

If the readings do not repeat, the rotor's unbalance is variable and the rotor cannot generally be balanced within tolerance until this problem is corrected.

E.3 Rotor flexibility test

Add a mass to the centre of the rotor, or to an available position where it may be expected to cause high rotor vibration. Accelerate the rotor to service speed, taking care that vibration at all times stays within safe limits. If the vibration magnitude becomes excessive during the acceleration, reduce the magnitude of the mass and repeat the process. Measure the vibration vector at service speed and at the same location as in E.2.3. Determine the effect of the mass on the vibration level by vectorially subtracting the vibration vector recorded in E.2.3 from the new reading. Denote the result by vector $\vec{A}$.

Stop the rotor and remove the mass. Install two masses at the same angular position as the central mass that was removed. These masses should be placed close to the rotor end planes. The masses should be chosen to provide the same quasi-static unbalance in the plane of the single test mass without introducing any additional couple unbalance. Accelerate the rotor to service speed again, take another reading and determine the effect of the two masses on the rotor by subtracting the vector from E.2.3 from the reading. Denote this vector by $\vec{B}$.

E.4 Evaluation of flexibility test data

Compute the magnitude of the vector $(\vec{A} - \vec{B})$. If this magnitude when divided by the magnitude of vector $\vec{A}$ is less than 0,2, the rotor can usually be considered rigid for balancing purposes. Conversely, if this ratio is 0,2 or greater then the rotor should be treated as a flexible rotor.

If sufficient rotor system modelling data are available, it is possible to generate analytically the data needed for calculating the ratio in E.4, thereby avoiding the need to perform the flexibility test. Particular care must be taken with this approach to model accurately the stiffness and damping characteristics of the rotor/support system.

Annex F
(informative)

Example — Permissible equivalent modal unbalance calculation (refers to 8.3.3.2)

Rotor	Turbo compressor
Quality grade	G2.5 from ISO 1940-1
Service speed	15 000 r/min
Rotor mass	1 000 kg

Assume low-speed balancing in two planes close to bearings.

Total residual unbalance for an equivalent rigid rotor according to ISO 1940:

$$1{,}60\ \frac{\text{g}\cdot\text{mm}}{\text{kg}} \times 1\,000\ \text{kg} = 1\,600\ \text{g}\cdot\text{mm}$$

Permissible equivalent first modal unbalance (60 %)	960 g·mm
Permissible equivalent second modal unbalance (60 %)	960 g·mm
Total residual unbalance for the rigid rotor	1 600 g·mm (800 g·mm per plane)

Annex G
(informative)

A method of computation of unbalance correction

The following is a suggested method of computation of unbalance correction by observation of the effect of a trial mass set.

Let vector $\overrightarrow{OA}$ in figure G.1 represent the initial vibration plotted to some arbitrary reference angle.

Let vector $\overrightarrow{OB}$ represent the resultant vibration, at the same speed and plotted to the same reference, when a trial mass set is added to the rotor.

Then the 'effect' of the trial mass set is represented in amplitude and angle by the vector $\overrightarrow{AB}$.

Therefore in order to nullify the original vibration, the trial mass set should be moved through the angle BAO and each mass in the set adjusted in size in the ratio $\frac{AO}{AB}$

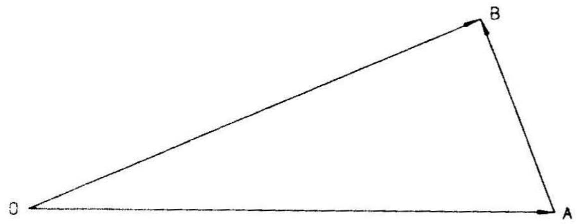

Figure G.1 — Vectorial effect of a trial mass set

Annex H

(informative)

Definitions from ISO 1925:1990 and ISO 1925:1990/Amd 1:1995 relating to flexible rotors

H.1 (rotor) flexural critical speed: Speed of a rotor at which there is maximum flexure of the rotor and where the flexure is significantly greater than the motion of the journals.

H.2 rigid-rotor-mode critical speed: Speed of a rotor at which there is maximum motion of the journals and where that motion is significantly greater than the flexure of the rotor.

H.3 (rotor) flexural principal mode: For undamped rotor/bearing systems, that mode shape which the rotor takes up at one of the (rotor) flexural critical speeds.

H.4 multiplane balancing: As applied to the balancing of flexible rotors, any balancing procedure that requires unbalance correction in more than two correction planes.

H.5 modal balancing: Procedure for balancing flexible rotors in which unbalance corrections are made to reduce the amplitude of vibration in the separate significant principal flexural modes to within specified limits.

H.6 mode function, $\phi_n(z)$: A mathematical expression for the deflection shape of the rotor in the corresponding mode.

H.7 modal mass, m_n: A scaling factor with dimensions of mass used in part to describe the mode function, expressed by

$$m_n = \int_0^L \mu(z)\, \phi_n^2(z)\, dz$$

where $\mu(z)$ is the mass per unit length of the rotor and L is the rotor length.

H.8 *n*th modal unbalance: That unbalance which affects only the *n*th principal mode of the deflection configuration of a rotor/bearing system.

NOTE 1 A measure of this component of unbalance is given by:

$$\vec{U}_n = \int_0^L \mu(z)\, \vec{e}(z)\, \phi_n(z)\, dz = \vec{e}_n m_n$$

where $e(z)$ is the eccentricity of the local mass centre at point z along the rotor.

NOTE 2 The *n*th modal unbalance is not a single unbalance but an unbalance distribution in the *n*th mode

$$\vec{u}_n(z) = \vec{e}_n\, \mu(z)\phi_n(z) = \frac{\vec{U}_n}{m_n}\, \mu(z)\phi_n(z)$$

It can be mathematically represented with respect to its effect on the *n*th principal mode by the single unbalance vector $\vec{U}_n$ as:

$$\int_0^L [\vec{e}_n\, \mu(z)\, \phi_n(z)]\phi_n(z)\, dz$$

$$= \vec{e}_n \int_0^L \mu(z)\, \phi_n^2(z)\, dz = \vec{e}_n m_n = \vec{U}_n$$

H.9 modal eccentricity (specific modal unbalance), *n*th mode: The *n*th modal unbalance divided by the *n*th modal mass

$$\vec{e}_n = \frac{\vec{U}_n}{m_n}$$

H.10 equivalent *n*th modal unbalance: The minimum single unbalance $\vec{U}_{ne}$, equivalent to the *n*th modal unbalance in its effect on the *n*th principal mode of the deflection configuration.

NOTES

1 There exists the relation $\vec{U}_n = \vec{U}_{ne}\, \phi_n(z_e)$ where $\phi_n(z_e)$ is the mode function value for $z = z_e$, the axial coordinate of the transverse plane where $\vec{U}_{ne}$ is applied.

2 A set of masses distributed in an appropriate number of correction planes and so proportioned that the mode under consideration will be affected, may be called the equivalent *n*th modal unbalance set.

3 An equivalent *n*th modal unbalance will affect some modes other than the *n*th mode.

H.11 modal balance tolerance: With respect to a mode, that amount of equivalent modal unbalance that is specified as the maximum below which the state of unbalance in that mode is considered to be acceptable.

H.12 multiple-frequency vibration: Vibration at a frequency corresponding to an integral multiple of the rotational speed.

NOTE — This vibration may be caused by anisotropy of the rotor, non-linear characteristics of the rotor/bearing system, or other causes.

H.13 thermally induced unbalance: That change in condition exhibited by a rotor if its state of unbalance is significantly altered by its changes in temperature.

NOTE — The change in condition may be permanent or temporary.

H.14 low-speed balancing (relating to flexible rotor): Procedure of balancing at a speed where the rotor to be balanced can be considered to be rigid.

H.15 high-speed balancing (relating to flexible rotors): Procedure of balancing at a speed where the rotor to be balanced cannot be considered to be rigid.

Annex I

(informative)

Bibliography

ISO 2953:1985, *Balancing machines — Description and evaluation*

ISO 7919 (all parts), *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria*

ISO 10814:1996, *Mechanical vibration — Susceptibility and sensitivity of machines to unbalance*

ISO 10816 (all parts), *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts*

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